

Experimental and Finite Element Analysis of Ball Screw Drive Vibrations

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Abstract

This study conveys the results of long stroke ball screw vibrations that were studied using a nonlinear finite element method. Because the nut in the experiment moved both forward and backward, a formulation that incorporated an updated Lagrangian method was employed to examine large displacement in the lateral directions. When the nut changed position, one of the screw segments became longer and its natural frequencies decreased, resulting in the formation of high-to-low frequency curves. However, when this occurred, the other screw segment became shorter and its natural frequencies increased, causing low-to-high frequency curves. A short-time Fourier transformation was used to determine the spectrograms for the forward and backward motions of the nut. Therefore, every frequency curve on the spectrograms that corresponded to one vibration mode formed a particular "ripple pattern." Notable vibration signatures were verified in laboratory tests that measured the time-frequency spectrum of a stationary screw with a moving nut. The results of different nut speeds were also gathered and the ripple patterns were identified as being independent of nut motion.

Keywords

Ball Screw; Nonlinear Finite Element Method; Vibration Ripple Patterns

Introduction

A ball screw drive is the most critical drive component in modern precision machines such as machine tools and aerospace, electronics, and health care devices. Using balls, screws, and nuts, the drive can convert rotational motion into linear motion. Moreover, because rolling replaces sliding, the friction coefficient decreases substantially, thereby enabling the initiation of motion using less torque, which enhances machine efficiency. However, during the motion process, impact and friction between various components

cause the ball screw drive to become a source of noise and vibration problems. Noise generation may subsequently influence worker comfort, and vibration can reduce machining precision and the life of the ball screw drive. With the development of the long stroke ball screw drive, solving vibration and noise problems has become increasingly critical. Thus, the establishment of simulation analysis methods for ball screw drive vibration and noise during initial design stages is an indispensable research and development tool for designers.

Among current analysis methods, the finite element method (FEM) has been widely adopted to address and resolve various engineering problems. For example, Holroyd, Pislaru, and Ford [1], Smith [2], and Erkorkaz and Kamalzadeh [3] have applied FEM to analyze the structure of ball screw drives, including examinations of axial, rotational, and lateral motion. Zaeh, Oertli, and Milberg [4] employed the stiffness of all investigated balls to establish the nut-screw coupling stiffness matrix, which could couple axial and rotational motion. Furthermore, Okwudire and Altintas [5] added lateral motion coupling to enable the generation of rotational, axial, and lateral motions using motor torque. Nevertheless, these studies have merely investigated traditional ball screw drive systems in which the screw is driven using a motor. Consequently, the screw is rendered an active rotational component and the nut becomes an entity involved only in passive rotational motion.

The mentioned scholars emphasized the relationship between lateral and axial forces generated by torque, and the nuts in their analysis models were all situated in a fixed position. Consequently, this study employed mechanism analysis to calculate the excitation force

generated by hundreds of balls during nut rotation. In addition, a three-dimensional (3D) nonlinear finite element vibration model was established to analyze the transient response activated by the ball screw drive under the effects of excitation force during the forward and backward motion of the nut. Finally, the spectrograms of transient responses were used to examine the distinct vibration ripple patterns produced by the nut.

The long stroke linear module proposed in this study was a nut-rotating or nut-rotation type ball screw, in which the screw was fixed and the nut demonstrated axial motion with rotation driven by a motor. Because the screw was fixed, the nut could move axially at a high speed to reduce motion time. The application of a large external diameter could also reduce the driving torque required by the motor. However, the stroke enlarged the ball screw and caused a larger deformation because of the weight of the screw. In addition, unbalanced nut rotation resulted in wobbling. Solutions to these problems included increasing the diameter of the screw to enhance stiffness and increasing the critical frequency to reduce deformation resulting from the weight of the screw. Nevertheless, these methods also increased overall weight and volume. Although increasing the preload of the screw can enhance stiffness, doing so also reduces screw life.

The long stroke ball screw drive vibration model was simulated using the nonlinear FEM, in which the screw, nut, returning tube, bearing, block, linear guide, motor, and cage were simulated using solid elements. To conserve computational resources, the screw was replaced by a screw without thread that possessed the same stiffness and weight. The velocity of motion of the nut was identical to that of the mechanism analysis and was affected by excitation forces from the mechanism kinematic analysis. Through contact, excitation force was transmitted to the screw. Moreover, this large displacement analysis assumed a small strain value and adopted a modified Newton-Raphson integration method.

Calculation of Effective Screw Diameter

A real screw possesses thread (Fig. 1). To conserve computational resources, an effective diameter screw without thread can be used as a substitute during mesh generation. However, the stiffness and weight of the screw without thread must be identical to those of the screw with thread.

A relatively short screw was first excised from the screw with thread to generate and delineate mesh. Subsequently, the cantilever beam boundary conditions were set. The three directions of the fixed end were all restrained with an external force exerted on the central point of the end surface. The screw was composed of steel, with a Young's modulus of $E = 2.11 \times 10^4 \text{ kg/mm}^2$, a Poisson ratio of $\nu = 0.3$, and a density of $\rho = 7850 \times 10^{-9} \text{ kg/mm}^3$. A static analysis of the beam was then conducted to determine the deformation. Afterward, the cantilever beam equation was used to calculate the area moment of inertia I before determining the effective diameter d_e .

The calculated effective diameter d_e was used to establish the model for the screw without thread (Fig. 2). During mesh generation, superior parabolic hexagonal elements can be used to conduct the generation process. When defining material attributes, although steel was used, because the effective diameter screw replaced the original screw with thread, the substituted screw possessed a smaller volume. Therefore, the density had to be adjusted to achieve identical weight between the two screws before conducting modal analysis.

A modal analysis of the original screw with thread (the material used for the screw was steel) was also performed. The mesh was composed of parabolic tetra elements, as shown in Fig. 1. The modal analysis results and experimental data of the screws with and without thread were compared, as shown in Table 1.

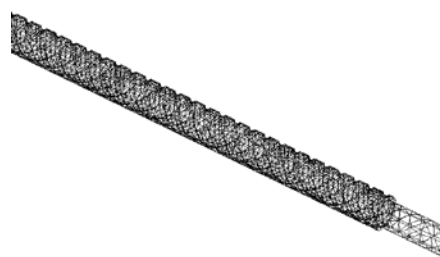


FIG. 1 FINITE ELEMENT MESH OF THE SCREW WITH THREAD

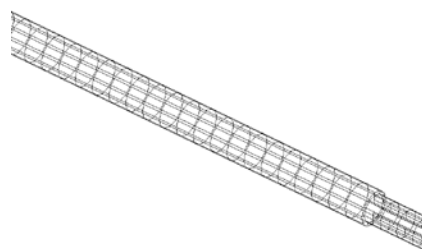


FIG. 2 FINITE ELEMENT MESH OF THE SCREW WITHOUT THREAD

TABLE 1 COMPARISON OF THE NATURAL FREQUENCIES DERIVED FROM TEST AND ANALYSIS RESULTS

Free-free beam	Test	FEM (without thread)	FEM (with thread)
Total nodes	-	69 229	219 760
Total elements	-	14 880	138 312
Element type	-	Hex20	Tetra10
Mode 1 (Hz)	16	14.7	14.8
Mode 2 (Hz)	40	40.2	40.5
Mode 3 (Hz)	78	78.4	78.9
Mode 4 (Hz)	127	128.9	129.7
Mode 5 (Hz)	189	191.5	192.7
Mode 6 (Hz)	263	266.0	267.5
Mode 7 (Hz)	349	352.1	353.8
Mode 8 (Hz)	446	449.5	451.5
Mode 9 (Hz)	554	558.0	560.4
Mode 10 (Hz)	674	677.3	679.9

After comparing the experimental data, the results showed that the effective-diameter screw without thread generated modal analysis results that were nearly identical to the experimental data, with an error of approximately 1%. Furthermore, the results for the screw without thread were more consistent with the experimental results than those for the screw with thread. Consequently, the results verified that, within a certain frequency range, the effective diameter method could be used to replace a screw with thread with a screw without thread.

Nonlinear Finite Element Model

The finite element vibration model of the long stroke linear module was composed of eight single parts (Fig. 3). The table, motor, bearing, block, and nut were all connected using rigid elements; the top panel of the cage and the caps of the front and rear ends were also connected using rigid elements; and the linear guide and the bottom panel of the cage were compatible meshes, which did not necessitate connection using rigid elements. In addition, the nut, screw, block, and linear guide employed deformable contact conditions. To drive the nut, a surface of the bearing cap, which formed rigid contact conditions, was glued to the nut. The motion condition was defined as a force behind the rigid contact surface that caused the nut to move or be driven, and the excitation forces from the mechanism analysis [6] were defined as being exerted onto the nut and employing the contact condition to transmit the force to the screw and the cage. The weight of each component was determined by adjusting the density of the finite element model based on actual weights. The Young's modulus represents the standard value of general aluminum alloys or steel materials (Table 2). The restraint condition was

delineated as existing at the bolt hole node of the bottom panel of the cage. Moreover, no translation degree of freedom existed for X, Y, or Z. The full vibration analysis model is shown in Fig. 3. The nonlinear finite element package used in this study was MSC.Marc [7].

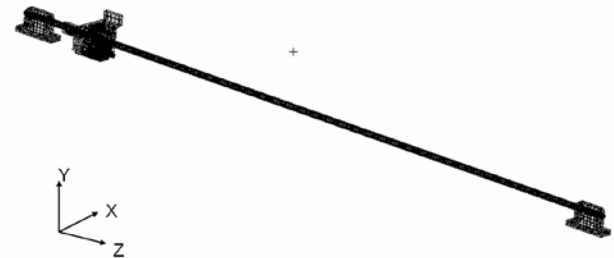


FIG. 3 FINITE ELEMENT MODEL FOR VIBRATION ANALYSIS

TABLE 2 PROPERTIES OF EACH COMPONENT

Component name	Density (kg·s ² /mm ⁴)	Young's modulus (kg/mm ²)
Nut	7.863e ⁻¹⁰	21100
Screw	8.925e ⁻¹⁰	21100
Bearing Cap	2.844e ⁻¹⁰	7342
Cage	2.753e ⁻¹⁰	7342
Motor	6.908e ⁻¹⁰	7342
Block	8.0045e ⁻¹⁰	21100
Linear guide	2.749e ⁻¹⁰	21100
Table	3.1505e ⁻¹⁰	7342

The nonlinear finite element equation employs the updated Lagrange method [8]. The configuration at time t is adopted as the reference configuration to determine the unknown variables of the structure at $t+\Delta t$. The matrix equation is shown below:

$$[M]^{t+\Delta t} \{\ddot{U}\}^{(k)} + [C]^{t+\Delta t} \{\dot{u}\}^{(k)} + t[K]\{\Delta U\}^{(k)} = t+\Delta t \{R\} - t+\Delta t \{F\}^{(k-1)} \quad (1)$$

where $t+\Delta t \{U\}^{(k)} = t+\Delta t \{U\}^{(k-1)} + \{\Delta U\}^{(k)}$

The modified Newton-Raphson method was used in (1) to determine relevant solutions. In other words, the tangent stiffness matrix was calculated only once. In each step, the same slope was employed to substantially reduce the calculation time. However, the convergence of each step was higher than the iteration numbers.

Flexible solid elements at one screw end and at the bearing housing were used to delineate contact conditions for the screw preload, and an axial load was further integrated. To facilitate rapid attainment of a steady state in the system at the point of screw preload, the preload end should include a dashpot

damper with a size of 1 kg•s/mm. In addition, the damper of the vibration analysis model was defined as Rayleigh damping; $[C] = \beta[K]$, where $\beta = 3.33\text{e-}5$.

Mode Frequencies

At 3000 rpm, the nut moved along the Z axis at a forward speed of 2000 mm/s. During analysis, this speed was defined as acting on rigid contact. Therefore, the nut was driven, and load was defined as being exerted on the nut. The analysis duration was 0.855 s. The preload reaction time accounted for the first 0.005 s, in which no external force was excited. The analysis time step was a fixed time step of $4.16667\text{e-}5$ s, and, consequently, a total of 20 520 increments were achieved. During the vibration analysis, the Z axis served as the nut motion direction, the X axis served as the horizontal and lateral directions, and the Y axis acted as the direction of the gravitational field ($g = 9807 \text{ mm/s}^2$). Fig. 4 shows that a Kistler trial-axial accelerometer was located near the right shoulder of the screw. This point also served as the screw response observation point (point P) for the finite element vibration analysis. When the nut moved away or was separated from point P (a movement from the right end to the left end), this motion was referred to as forward motion, and a reverse motion from the left end to point P at the right was referred to as backward motion. During the forward and backward motions, the time-frequency transient vibration of point P was converted into a spectrogram using the short-time Fourier transform (STFT) [9]. Fig. 5 demonstrates the spectrograms for point P and shows that during forward motion, the screw length at the right segment increased because the nut moved toward the left (away from point P), and the natural frequencies of the vibration decreased with time. In contrast, during backward motion, the screw length at the right segment decreased because the nut moved toward the right (and in the direction of point P), and the natural frequencies of the vibration increased with time. The natural frequencies of the screw length at the right segment can be expressed as follows:

$$\frac{2\beta^3(\psi_1 - 1)}{\psi} \left\{ \frac{\beta(-m + \frac{k}{\omega^2})}{A\rho} - \left[\frac{(\psi_2 + \psi_3)}{(1 - \psi_1)} + \frac{(\varphi_2 + \varphi_3)}{(1 - \varphi_1)} \right] \right\} = 0 \quad (2)$$

where

$$\psi = \cos(L_2\beta) - \cosh(L_2\beta) \quad (3)$$

$$\psi_1 = \cos(L_2\beta) \cosh(L_2\beta) \quad (4)$$

$$\psi_2 = \sin(L_2\beta) \cosh(L_2\beta) \quad (5)$$

$$\psi_3 = \cos(L_2\beta) \sinh(L_2\beta) \quad (6)$$

$$\varphi_1 = \cos(L_1\beta) \cosh(L_1\beta) \quad (7)$$

$$\varphi_2 = \sin(L_1\beta) \cosh(L_1\beta) \quad (8)$$

$$\varphi_3 = \cos(L_1\beta) \sinh(L_1\beta) \quad (9)$$

where L_1 is the length of the screw at the right segment and L_2 is the length of the screw at the left segment (Fig. 4). For beams with two fixed ends, the first, second, third, and fourth natural frequency coefficients [10] are $\beta_1 L = 4.730041$, $\beta_2 L = 7.853205$, $\beta_3 L = 10.995608$, $\beta_4 L = 14.137165$, respectively.

Fig. 5 indicates that the natural frequencies of vibration for the screw at the right segment increased and decreased due to nut motion. The simulation analysis results also showed that changing the nut rotational speed only altered the excitation force exerted on the screw. The increase or decrease of the time axis did not influence the spectrograms. Figs. 6 and 7 respectively show the measurement results of point P at the nut rotational speeds of 3000 and 1000 rpm. The experimental results correspond to the computer simulation analysis predictions.

In the vibration analysis model and at 3000 rpm, the nut moved at a fixed speed ($v = 2000 \text{ mm/s}$) for 0.85 s, resulting in a 1.7 m stroke. During the practical test, the nut had to perform accelerating, fixed-speed, and decelerating motions to complete a 1.7 m stroke. Therefore, the lengths of the time axes in Figs. 5 and 6 differed slightly. Carefully comparing the backward motions (3000 rpm) in Figs. 5 and 6 illustrated that both the analysis and measured values were 1000 Hz when the nut moved to the right end. This conveys consistency between the computer simulation analysis and the measurement results. Regarding the backward motion (1500 rpm) shown in Fig. 7, when the nut moved to the right end, the first natural frequency was maintained at 1000 Hz. Thus, the characteristics of ripple patterns in the spectrograms were not correlated with nut rotational speed.

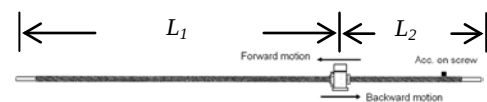


FIG. 4 TEST SET-UP FOR THE LONG STROKE BALL SCREW SYSTEM

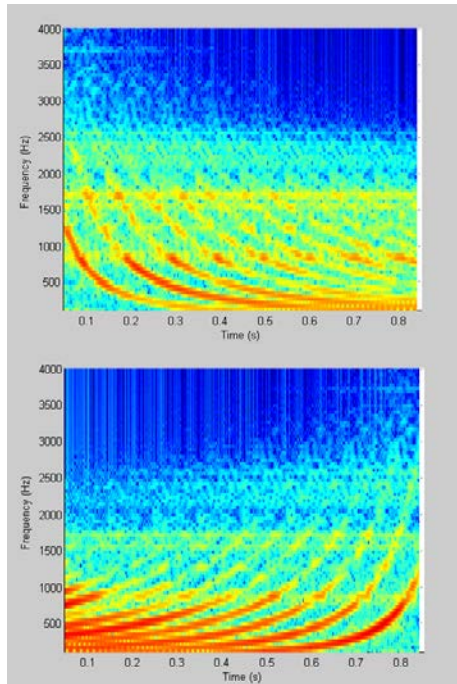


FIG. 5 ANALYSIS RESULTS FOR THE SCREW SPECTROGRAMS. (A) FORWARD, (B) BACKWARD

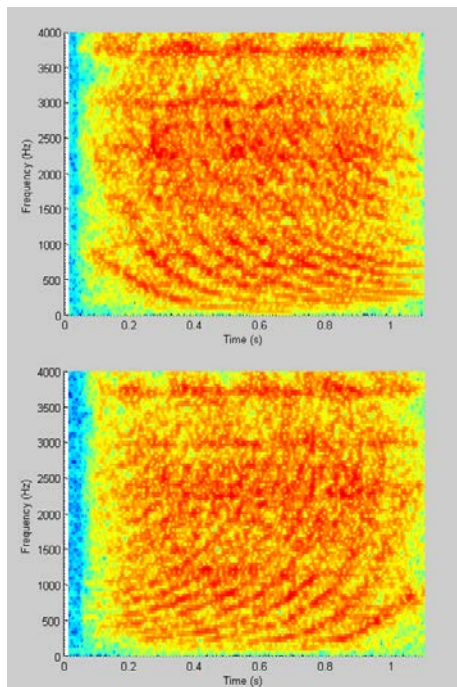


FIG. 6 TEST RESULTS FOR THE SCREW SPECTROGRAMS (3000 RPM). (A) FORWARD, (B) BACKWARD

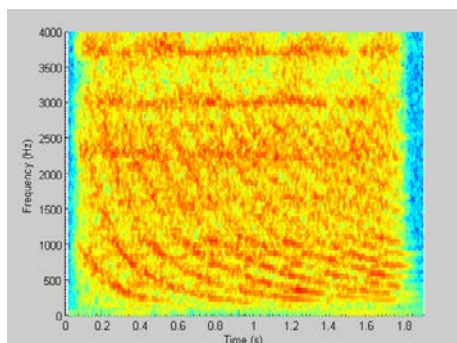


FIG. 7 TEST RESULTS FOR THE SCREW SPECTROGRAMS (1500 RPM). (A) FORWARD, (B) BACKWARD

Conclusion

This study adopted the nonlinear FEM to simulate the vibration behavior of a long stroke ball screw drive driven by a nut. The research results indicated that the forward and backward motions of the nut and the differing positions of the nut led to the varying lengths of the right and left segments of the screw. During forward motion, the screw segment at the right became longer, thereby decreasing the natural frequencies. In contrast, during backward motion, the screw segment at the right became shorter, thereby increasing the natural frequencies. The transient signals of the observation point were converted into spectrograms using STFT. The spectrograms demonstrated that during forward motion, all modal natural frequencies decreased with time, whereas the natural frequencies during a backward motion increased with time. The two spectrograms demonstrated ripple pattern characteristics that did not change with the varying rotational speeds of the nut.

The time at which the ball screw drive passes critical speed is transient under a fixed rotational speed. Consequently, if vibration is sustained only for a short period of time, the screw can operate at a speed higher than the critical speed. However, because the long stroke screw possesses low radial stiffness, vibration attenuation cannot be easily attained, thereby resulting in vibration problems for other components. A solution to this issue is to increase the dashpot damper or change the support method, which are both directions that can be adopted in future studies.

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